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Original Article

IMPACT OF INJECTION TIMING AND INJECTION PRESSURE ON PERFORMANCE PARAMETERS AND COMBUSTION CHARACTERISTICS OF HIGH GRADE SEMI ADIABATIC DIESEL ENGINE WITH COTTON SEED BIODIESEL

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ABSTRACT

The biodiesel available from vegetable oil feedstock is an important substitutes for diesel fuel, due to comparable properties with diesel fuel. It is easily available and renewable in nature. However, drawbacks associated with biodiesel of moderate viscosity and low volatility call for semi adiabatic diesel engine. The objective of semi adiabatic diesel engine is to minimize heat loss to the coolant, by providing thermal resistance to the heat flow to the coolant. In this work cottonseed biodiesel was used as sole fuel in conventional diesel engine and LHR direct injection (DI) diesel engine. It consisted of an air gap insulated piston, an air gap insulated liner and ceramic coated cylinder head with different operating conditions of cotton seed biodiesel with varied injection timing and injector opening pressure. Combustion characteristics were determined at full load operation with special pressure-crank angle software package. Performance parameters of brake thermal efficiency, brake specific energy consumption, exhaust gas temperature, coolant load and volumetric efficiency were determined. At full load operation of the engine. Combustion characteristics of peak pressure, maximum rate of pressure rise and time of occurrence of peak pressure were evaluated at full load operation of both versions of the engine. LHR engine with biodiesel increased peak brake thermal efficiency by 3% at manufacturer's recommended injection timing and 10% at optimum injection timing in comparison with conventional engine with biodiesel operation at recommended injection timing and optimum injection timing. Semi adiabatic engine fuelled with biodiesel showed improved performance and combustion characteristics at 27^o bTDC and at optimum injection timing over CE.

KEYWORDS: ***Vegetable Oil, Biodiesel, LHR Combustion Chamber, Fuel performance,Combustion Characteristics***

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INTRODUCTION

Due to its fuel economy the use of diesel engine is agriculture sector and transport sector is gaining prominence day by day and with this, its consumption is higher than that of gasoline fuel. In the scenario of fast exhausting of fossil (diesel) fuels. search for renewable fuels is necessary. Vegetable oils can be selected as alternative fuels for diesel engines, as they are renewable and posses same properties as that of diesel. Rudolph Diesel, carried out investigations successfully on diesel engine with peanut oil and predicted that vegetable oils could be successfully used as alternative fuels. Several researchers carried out investigations with vegetable oils as fuel on conventional engines (CE). They reported that the performance with conventional engine was poor, because of low volatility, high viscosity and presence

of fatty acids. It caused injection problems, combustion chamber deposit problems, decreased power, increased fuel consumption and increased exhaust gas pollutnats. [1–5].

However, the above mentioned problems of crude vegetable oil can be rectified so some extent, by converting crude vegetable oils to biodiesel by the process known as esterification. Investigations were carried out with biodiesel on CE to evaluate the performance, exhaust emissions and combustion characteristics and compared with normal vegetable oil [6–10]. They reported from their studies that biodiesel operation on conventional engine showed marginal increase of brake thermal thermal efficiency, reduced particulate emissions and increased nitrogen oxide (NO_x) levels, in comparison with diesel operation on same version of the engine. .

Preheating temperature is a temperature at which viscosity of vegetable oil or biodiesel is matched to that of diesel fuel, to ensure there was no effect of viscosity of fuel on combustion. Experiments were conducted on engine with preheated vegetable oils. [11-13]. They reported that preheated vegetable oils increased thermal efficiency slightly, reduced particulate matter emissions and NO_x levels, in comparison with normal vegetable oil or biodiesel..

The performance of the engine can be increased with an increase of injection pressure. Investigations were carried out on engine with biodiesel with increased injector opening pressure. [14-15]. They reported that with an increase of injector opening pressure, marginal increase of performance of the engine, reduction of particulate emissions and increase of NO_x levels were observed.

The disadvantages with biodiesel (moderate viscosity and low volatility) aim for hot combustion chamber, provided by semi adiabatic diesel engine. The concept of the semi adiabatic diesel engine is to provide thermal resistance to the coolant thereby minimize heat loss to the coolant. Three methods that are being studied to decrease heat rejection are (1) thermal coating with low thermal conductivity materials on engine components of combustion chamber, (low semi adiabatic diesel engine); (2) air gap insulation with low thermal conductivity materials as inserts (medium grade semi adiabatic diesel engine); and (3).high grade semi adiabatic diesel engine contains air gap insulation and thermal coating.

Investigations were carried out on engine with high grade semi adiabatic diesel engine with biodiesel. It consisted of an air gap insulated piston, air gap insulated liner and ceramic coated cylinder head with varied injection timing and pressure with biodiesel. [16–22]. They reported from their studies, that engine with high grade semi adiabatic diesel engine with biodiesel increased brake thermal efficiency by 10–12%, at full load operation–decreased particulate emissions by 45–50% and increased NO_x levels, by 45–50% at an optimum injection timing of 28^o bTDC when compared with diesel operation on conventional engine at 27^o bTDC.

The present paper attempted to determine the performance of the engine with high grade semi adiabatic diesel engine. It contained an air gap (3.2 mm) insulated piston with superni (an alloy of nickel) crown, an air gap (3.2 mm) insulated liner with superni insert and ceramic coated cylinder head with cotton seed biodiesel with varied injection timing and injector opening pressure. Results were compared with CE with biodiesel and also with diesel at similar operating conditions.

MATERIALS AND METHODS

Cottonseeds have approximately 18% (w/w) oil content. [8]. India’s cottonseed production is estimated to be around 35% of its cotton output (approximately 4.5millionmetric tons). [8]. Approximately 0.30 million metric ton cottonseed oil is produced in India and it is an attractive biodiesel feedstock [8]

Preparation of Biodiesel

The preparation of biodiesel was mentioned in Ref [25].The properties of the Test Fuels used in the experiment were presented in Table 1. [8].

Table 1: Properties of Test Fuels [8]

Property	Units	Diesel (DF)	Biodiesel(BD)	ASTM Standard
Carbon Chain	--	C_8–C_{28}	C_{16}–C_{24}	---
Cetane Number	-	51	56	ASTM D 613
Specific Gravity at 15°C	-	0.8275	0.8673	ASTM D 4809
Bulk Modulus at 15°C	MPa	1408.3	1564	ASTM D 6793
Kinematic Viscosity @ 40°C	cSt	2.5	5.44	ASTM D 445
Air Fuel Ratio (Stoichiometric)	--	14.86	13.8	--
Flash Point (Pensky Marten's Closed Cup)	°C	120	144	ASTM D93
Cold Filter Plugging Point	°C	Winter 6° C Summer 18°C	3° C	ASTM D 6371
Pour Point	°C	Winter 3°C Summer 15°C	0°C	ASTM D 97
Sulfur	(mg/kg, max)	50	42	ASTM D5453
Low Calorific Value	MJ/kg	42.0	39.9	ASTM D 7314
Oxygen Content	%	0.3	11	--

Engine with LHR Combustion Chamber

The assembly details of air gap insulated piston, air gap insulated liner and ceramic coated cylinder head were given in Ref [25].

Experimental Set–Up

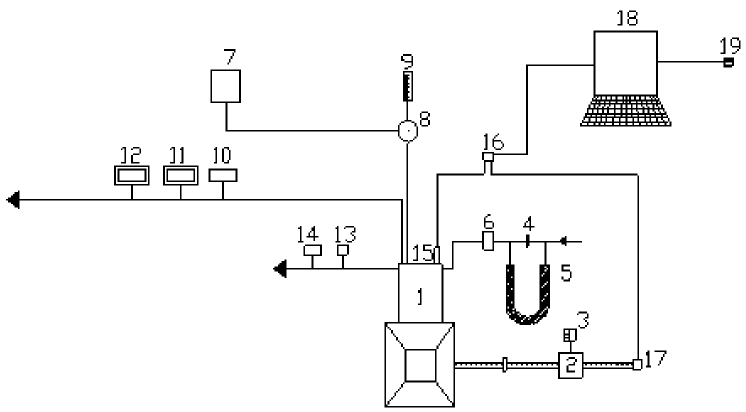

Figure 1: Shows the Schematic Diagram of the Experimental Setup Used for the Investigations on the Engine with High Grade Semi Adiabatic Diesel Engine with Cotton Seed Biodiesel

1.Four Stroke Kirloskar Diesel Engine, 2.Kirloskar Electical Dynamometer, 3.Load Box, 4.Orifice flow meter, 5.U-tube water manometer, 6.Air box, 7.Fuel tank, 8, Pre-heater 9.Burette, 10. Exhaust gas temperature indicator, 11.AVL Smoke opacity meter,12. Netel Chromatograph NO_x Analyzer, 13.Outlet jacket water temperature indicator, 14. Outlet-jacket water flow meter, 15.AVL Austria Piezo-electric pressure transducer, 16.Console, 17.AVL Austria TDC encoder, 18.Personal Computer and 19. Printer.

Table 2 gives Specifications of Test engine. Dynamometer used for measuring brake power of the engine was coupled to engine. accuracy of measurement of ±1%. Load box arrangement was provided to load the engine. Burette methods was used to measure fuel consumption of the engine, while air box method was used for measuring air consumption of the engine. The outlet temperature of water was maintained at 80^{o}C by adusting water flow rate. Pressure feed system was provided for engine oil. There was no facility incorporated to measure temperature of lubricating oil temperature. In order to change injection timing, copper shims of suitable size were provided in between the engine frame and pump body. By means of nozzle testing device, nozzle injection pressure was changed from 190 to 270 bar. De to practical difficulties involved, maximum injector opening pressure was restricted to 270 bar. Exhaust gas temperature, coolant water jacket inlet temperature, outlet water jacket temperature were measured by employing iron and iron-constantan thermocouples connected to analogue temperature indicators. Piezo electric transducer, (AVL (Austria: QC34D). connected to cylinder head, which in turn was connected to console. TDC encoder (AVL Austria: 365x) connected to extended portion of shaft of dynamometer, which in turn was connected to console. Console was connected to computer.

Table 2: Specifications of Test Engine

Description	Specification
Engine make and model	Kirloskar (India) AV1
Maximum power output at a speed of 1500 rpm	3.68 kW
Number of cylinders ×cylinder position× stroke	One × Vertical position × four-stroke
Bore × stroke	80 mm × 110 mm
Engine Displacement	553 cc
Method of cooling	Water cooled
Rated speed (constant)	1500 rpm
Fuel injection system	In-line and direct injection
Compression ratio	16:1
BMEP @ 1500 rpm at full load	5.31 bar
Manufacturer's recommended injection timing and injector opening pressure	27^{o}bTDC × 190 bar
Number of holes of injector and size	Three × 0.25 mm
Type of combustion chamber	Direct injection type

From the signals of pressure and crank angle, crank angle diagram was obtained. The accuracy of measurement of pressure is ±1 bar, while it is ± 1^{o} for crank angle. Combustion parameters such as maximum rate of pressure rise, time of occurrence of peak pressure and peak pressure at the full load operation of the engine were determined.

RESULTS AND DISCUSSIONS

The optimum injection timing with CE was 31^{o} bTDC, while it was 28^{o} bTDC for engine with LHR combustion chamber with diesel operation [23-24]. Similarly the corresponding angles for biodiesel operation were 31^{o} bTDC and 28^{o} bTDC. [25].

Figure 2 presents bar charts showing the variation of peak BTE with test fuels. From Figure 2 it is observed that engine with LHR combustion chamber with biodiesel operation increased peak BTE by 3% at 27^{o} bTDC and 10% at 28^{o} bTDC in comparison with CE with biodiesel operation at 27^{o} bTDC and at 31^{o} bTDC. Improved evaporation of biodiesel in hot environment provided by the engine with LHR combustion chamber might have improved peak thermal efficiency of the engine. Engine with LHR combustion chamber with biodiesel operation showed higher peak BTE than diesel operation on same configuration of the engine.

This showed that engine with LHR combustion chamber was more suitable for biodiesel.

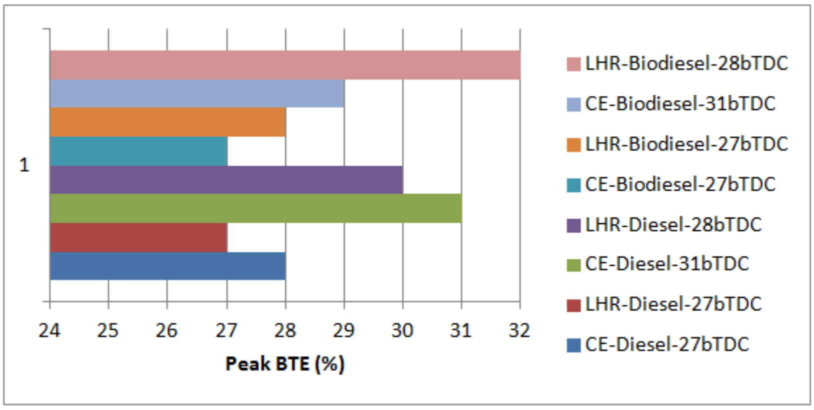

Figure 2: Bar Charts Showing the Variation of Peak Brake Thermal Efficiency (BTE) with Test Fuels with Conventional Engine (CE) and Engine with LHR Combustion Chamber at Recommended and Optimized Injection Timings at an Injector Opening Pressure of 190 Bar

Figure 3 presents bar charts showing the variation of brake specific energy consumption (BSEC) at full load with test fuels. From Figure 3, it is shown that BSEC at full load decreased with advanced injection timing with test fuels. This was because of increase of resident time of fuel with air thus improving atomization and thus combustion. BSEC was comparable with biodiesel with CE at 27^{o} bTDC and 31^{o} bTDC, when compared with CE with diesel operation at 27^{o} bTDC and at 31^{o} bTDC. Improved combustion with higher cetane number and presence of oxygen in fuel composition with higher heat release rate with biodiesel may lead to produce comparable BSEC at full load. Engine with LHR combustion chamber with biodiesel decreased BSEC at full load operation by 6% at 27^{o} bTDC and 3% at 28^{o} bTDC, when compared diesel operation with engine with LHR combustion chamber at 27^{o} bTDC and at 28^{o} bTDC. This once again confirmed that engine with LHR combustion chamber was more suitable for biodiesel operation than neat diesel operation. Engine with LHR combustion chamber with biodiesel decreased BSEC at full load operation by 3% at 27^{o} bTDC and 2% at 28^{o} bTDC, in comparison with CE at 27^{o} bTDC and at 31^{o} bTDC with biodiesel.

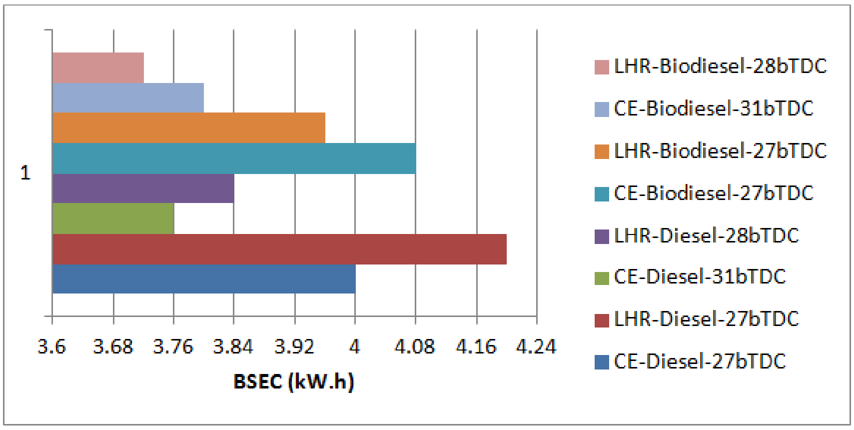

Figure 3: Bar Charts Showing the Variation of Brake Specific Energy Consumption (BSEC) at Full Load Operation with Test Fuels with Both Versions of the Engine at Recommended and Optimized Injection Timings at an Injector Opening Pressure of 190 Bar

Improved evaporation rate and higher heat release rate of fuel with LHR combustion chamber might have improved the performance with LHR engine.

Figure 4 presents bar charts showing variation of exhaust gas temperature (EGT) at full load with test fuels. From Figure 4, it is noticed that, exhaust gas temperature (EGT) at full load operation decreased with advanced injection timing with test fuels. This was because, gases expand at larger injection timing leading to decrease EGT as expansion stroke was increased. CE with biodiesel operation increased EGT at full load operation by 6% at 27^{o} bTDC and 7% at 31^{o} bTDC in comparison with CE with neat diesel operation at 27 ^{o} bTDC and at 31^{o} bTDC. Though calorific value (or heat of combustion) of biodiesel is lower than that of diesel, density of biodiesel is higher, therefore greater amount of heat was released in the combustion chamber leading to produce higher EGT at full load operation with biodiesel operation than neat diesel operation. This was also because of higher duration of combustion of biodiesel causing retarded heat release rate. Similar findings were obtained by other researchers [6–8]. From Figure 4, it is noticed that engine with LHR combustion chamber with biodiesel operation increased EGT at full load operation by 5% at 27^{o} bTDC and 5% at 28^{o} bTDC, when compared with diesel operation on same configuration of the engine at 27^{o} bTDC and at 28^{o} bTDC. High duration of combustion due to high viscosity of biodiesel in comparison with diesel might have increased EGT at full load. Engine with LHR combustion chamber with biodiesel increased EGT at full load operation by 17% at 27^{o} bTDC and 13% at 28^{o} bTDC, in comparison with CE at 27^{o} bTDC and at 31^{o} bTDC. This indicated that heat rejection was restricted through the piston, liner and cylinder head, thus maintaining the hot combustion chamber as result of which EGT at full load operation increased with reduction of ignition delay. Similar observations were reported by previous researchers [21–22].

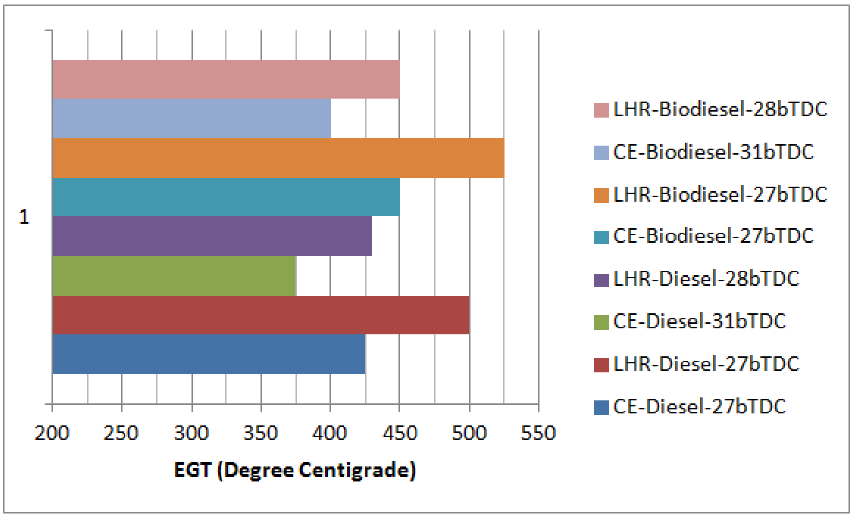

Figure 4: Bar Charts Showing the Variation of Exhaust Gas Temperature (EGT) at Full Load Operation with Test Fuels with Both Versions of the Engine at Recommended and Optimized Injection Timings at an Injector Opening Pressure of 190 Bar

Table 3 shows performance parameters of peak BTE, BSEC at full load and EGT at full load

Table 3: Comparative Data on Peak Brake Thermal Efficiency, Brake Specific Energy Consumption and Exhaust Gas Temperature at Full Load

IT/ Combustion Chamber Version	Test fuel	Peak Brake Thermal Efficiency (%)		Brake Specific Energy Consumption at Full Load Operation (kW.h)		Exhaust Gas Temperature (^{o}C) at Full Load Operation	
		Injector Opening Pressure (Bar)		Injector Opening Pressure (Bar)		Injector Opening Pressure (Bar)	
		190	270	190	270	190	270
27(CE)	DF	28	30	4.0	3.84	425	395
	BD	27	29	4.08	4.0	450	425
27(LHR)	DF	27	29	4.2	4.12	500	450
	BD	28	30	3.96	3.88	525	475
28(LHR)	DF	30	31.5	3.84	3.76	430	390
	BD	32	33	3.72	3.64	450	400
31(CE)	DF	31	31	3.76	3.84	375	325
	BD	29	30.5	3.80	3.72	400	350

From Table 3, it is noticed that preheating of the biodiesel improved the performance in both versions of the combustion chamber when compared with the biodiesel at normal temperature. Preheating reduced the viscosity of the biodiesel, causing efficient combustion thus improving BTE. Peak BTE improved marginally with an increase of injector opening pressure in both versions of the combustion chamber with test fuels. As injector opening pressure increased, droplet diameter decreased influencing the atomization quality, and more dispersion of fuel particle, resulting in enhanced mixing with air, leads to improved oxygen-fuel mixing rate, as extensively reported in the literature [13–15].

From Table 3, it is observed that BSEC at full load operation decreased marginally with an increase of injector opening pressure. This was because of improved heat release rate with reduction of size of fuel particle. BSEC at full load operation decreased with preheating of biodiesel in both versions of the combustion chamber. Improved spray characteristics with the reduction of viscosity of biodiesel might have reduced BSEC at full load with test fuels. From Table 3, it is noticed that EGT at full load operation increased marginally with preheated biodiesel with CE, With increase of fuel temperature, there was reduction of ignition delay and increase of diffusion combustion causing an increase of EGT. However, EGT at full load decreased marginally with engine with LHR combustion with preheated biodiesel due to improved combustion. EGT at full load reduced marginally with an increase of injector opening pressure in both versions of the combustion chamber as seen from Table 3. Improved combustion with improved oxygen–fuel ratios might have reduced EGT at full load with test fuels.

Figure 5 presents bar charts showing variation of coolant load with test fuels. CE with biodiesel increased coolant load by 3% at 27^{o} bTDC and 10% at 31^{o} bTDC when compared with neat diesel operation on CE at 27^{o} bTDC and 31^{o} bTDC as observed from Figure 5.

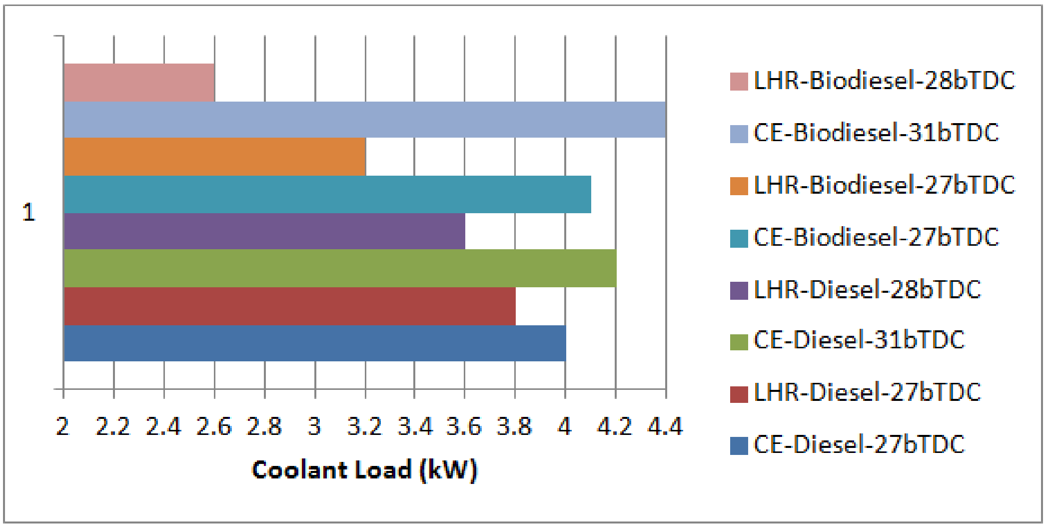

Figure 5: Bar Charts Showing the Variation of Coolant Load at Full Load Operation with Test Fuels with Both Versions of the Engine at Recommended and Optimized Injection Timings at an Injector Opening Pressure of 190 Bar

Increase of un–burnt fuel concentration at the combustion chamber walls may lead to increase of gas temperatures with biodiesel produced higher coolant load than diesel operation. Similar trends were reported in previous studies [21–22]. Coolant load at full load operation increased in CE, while decreasing the same in engine with LHR combustion chamber with advanced injection timing with biodiesel. For CE, un burnt fuel concentration at combustion chamber walls, decreased with combustion of fuel, leading to increase of coolant load, with an increase of gas temperatures, when the injection timing was advanced to the optimum value.. The reduction of coolant load in engine with LHR combustion chamber might be due to the reduction of gas temperatures with improved combustion. Hence, the improvement in the performance of CE was due to heat addition at higher temperatures and rejection at lower temperatures, while the improvement in the efficiency of the engine with LHR combustion chamber was because of recovery from coolant load at their optimum injection timings with test fuels. Engine with LHR combustion chamber with biodiesel operation decreased coolant load operation by 16% at 27^{o} bTDC and 28% at 28^{o} bTDC, when compared diesel operation with same configuration of the engine at 27^{o} bTDC and at 28^{o} bTDC. More conversion of heat into useful work with biodiesel than diesel might have reduced coolant load with biodiesel. Figure 5 indicates that engine with LHR combustion chamber with biodiesel decreased coolant load at full load operation by 7% at 27^{o} bTDC and 41% at 28^{o} bTDC, in comparison with CE at 27^{o} bTDC and at 31^{o} bTDC. Provision of thermal insulation and improved combustion with engine with LHR combustion chamber might have reduced coolant load with LHR engine in comparison with CE with biodiesel operation. Similar observations were reported by previous researchers. [21–22].

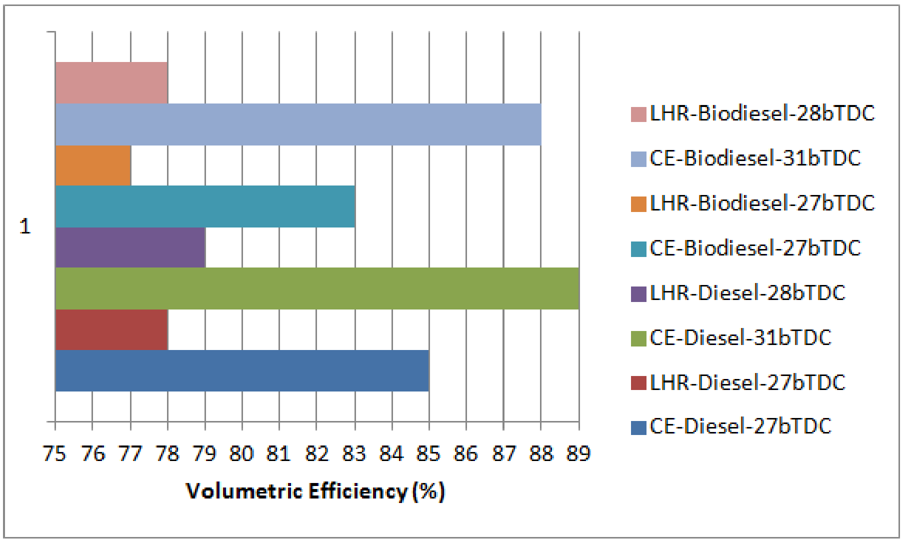

Figure 6: Shows Bar Charts Showing Variation of Volumetric Efficiency at Full Load with Test Fuels

Figure 6 Bar charts showing the variation of volumetric efficiency at full load operation with test fuels with both versions of the engine at recommended and optimized injection timings at an injector opening pressure of 190 bar.

It indicates that CE with biodiesel operation decreased volumetric efficiency at full load by 2% at 27^o bTDC and comparable at 31^o bTDC, when compared with diesel operation on CE at 27^o bTDC and 31^o bTDC. Increase of EGT might have reduced volumetric efficiency at full load, as volumetric efficiency depends on combustion wall temperature, which in turn depends on EGT. Volumetric efficiency at full load operation improved marginally with advanced injection timing with test fuels with both configurations of the combustion chamber. Reduction of EGT at full load might have improved volumetric efficiency with test fuels. From Figure 9, it is noticed that volumetric efficiencies at full load operation on engine with LHR combustion chamber at 27^o bTDC and at 28^o bTDC with biodiesel were marginally lower than diesel operation on same configuration of the engine at 27^o bTDC and 28^o bTDC. Increase of EGT was responsible factor for it. Figure 7 indicates that engine with LHR combustion chamber with biodiesel decreased volumetric efficiency at full load operation by 7% at 27^o bTDC and 11% at 28^o bTDC, in comparison with CE at 27^o bTDC and at 31^o bTDC. The reduction of volumetric efficiency with engine with LHR combustion chamber was because of increase of temperatures of insulated components of LHR combustion chamber, which heat the incoming charge to high temperatures and consequently the mass of air inducted in each cycle was lower. Similar observations were noticed by earlier researchers [21–22].

Table 4 shows coolant load and volumetric efficiency at full load. Coolant load at full load operation decreased with preheating of biodiesel, as noticed from Table 4. Improved spray characteristics might have reduced gas temperatures and hence coolant load. From Table 4, it is observed that that coolant load at full load operation marginally increased in CE, while decreasing it in engine with LHR combustion chamber with increased injector opening pressure with test fuels. This was due to the fact that increased injector opening pressure increased nominal fuel spray velocity resulting in improved fuel–air mixing with which gas temperatures increased in CE.

Table 4: Comparative Data on Coolant Load & Volumetric Efficiency at Full Load Operation

IT/ Combustion Chamber Version	Test fuel	Coolant Load (kW)		Volumetric Efficiency (%)	
		Injector Opening Pressure (Bar)		Injector Opening Pressure (Bar)	
		190	270	190	270
27(CE)	DF	4.0	4.4	85	87
	BD	4.1	4.5	83	85
27(LHR)	DF	3.8	3.4	78	80
	BD	3.2	2.8	77	79
28(LHR)	DF	3.6	3.0	79	81
	BD	2.6	2.2	78	80
31(CE)	DF	4.2	4.6	89	87
	BD	4.4	4.8	88	90

Improved fuel spray characteristics and increase of oxygen fuel rations might have reduced coolant load with an increase of injection pressure with semi adiabatic engine with biodiesel operation. Volumetric efficiency at full load operation marginally reduced in CE, while increasing it in engine with LHR combustion chamber with preheated biodiesel as observed from Table 4.

This was because of increase of EGT in CE, while decreasing of the same in engine with LHR combustion chamber. Volumetric efficiency at full load operation was marginally increased with an increase of injector opening pressure in both versions of the combustion chamber with test fuels. Improved oxygen–fuel ratios might have reduced EGT with test fuels. . However, these variations were very small.

Table 5 shows combustion characteristics at full load with test fuels. From Table 5. , it is noticed that CE with biodiesel increased peak pressure (PP) at full load operation by 4% at 27^o bTDC and 5% at 31^o bTDC when compared with diesel operation on CE at27^o bTDC and at 31^o bTDC. Even though biodiesel has lower heat of combustion, it advanced its peak pressure position because of its higher bulk modulus and cetane number. Peak pressure at full load operation with biodiesel increased with CE, while marginally decreasing it in engine with LHR combustion chamber with advanced injection timings. Increase of ignition delay of fuel in CE, causing sudden increase of pressures with accumulated fuel with advanced injection timings increased peak pressure in CE. Improved combustion with improved air–fuel ratios decreased peak pressure at full load operation on engine with LHR combustion chamber. Table 5 indicates that engine with LHR combustion chamber with biodiesel increased peak pressure at full load operation by 35% at 27^o bTDC and 5% at 28^o bTDC, in comparison with CE at 27^o bTDC and at 31^o bTDC. Improved heat release rate with engine with LHR combustion chamber might have increased peak pressure with LHR engine with biodiesel. Peak pressure at full load operation increased in CE, while marginally reducing it with engine with LHR combustion chamber with an increase of injector opening pressure with test fuels. Smaller sauter mean diameter shorter breakup length, better dispersion, and better spray and atomization characteristics might be responsible factors to increase peak pressure with CE with biodiesel. This improves combustion rate in the premixed combustion phase in CE. Improved combustion with improved air–fuel ratios marginally reduced peak pressure at full load operation with engine with LHR combustion chamber.

Table 5: Data of Combustion Characteristics at Full Load Operation

IT/ Combustion Chamber Version	Test Fuel	Peak Pressure (Bar)		Maximum Rate of Pressure Rise (Bar/Degree)		Time of Occurrence of Peak Pressure (Degrees)	
		Injector Opening Pressure (Bar)		Injector Opening Pressure (Bar)		Injector Opening Pressure (Bar)	
		190	270	190	270	190	270
27(CE)	DF	50.4	53.5	5.4	6.0	9	9
	BD	52.5	55.5	4.9	5.2	8	7
27(LHR)	DF	66.5	60.6	7.6	6.8	10	9
	BD	70.8	67.6	6.8	6.3	7	6
28(LHR)	DF	63.5	59.5	7.2	6.4	9	8
	BD	68.5	63.6	6.5	5.7	6	5
31(CE)	DF	62.4	60.6	6.2	5.8	8	9
	BD	65.4	62.6	5.6	6.0	7	8

Table 5 denotes that maximum rate of pressure rise (MRPR) was higher for diesel than biodiesel with both versions of the combustion. High volatile nature of diesel fuel releasing more energy per unit crank angle might have increased MRPR at full load. MRPR at full load operation followed similar trends at different operating conditions as in case of PP at full load operation. Engine with LHR combustion chamber increased MRPR at full load operation by 39% at 27^{o} bTDC and 16% at 28^{o} bTDC with engine, when compared with CE at 27^{o} bTDC and at 31^{o} bTDC with biodiesel, which showed that combustion improved with improved air–fuel ratios in hot environment provided by the engine with LHR combustion chamber.

AT recommended injection timing and optimum injection timing, biodiesel operation marginally decreased time of occurrence of peak pressure (TOPP) at full load operation on CE, while drastically decreasing it with engine with LHR combustion chamber, when compared diesel operation, as noticed from Table 5. Higher bulk modulus of rigidity and cetane number of biodiesel, in comparison with neat diesel operation might have reduced TOPP at full load operation. TOPP at full load operation decreased (towards TDC) with the advanced injection timing at different operating conditions of biodiesel with both versions of the combustion chamber as noticed from Table 5, which showed that combustion improved with advanced injection timing. TOPP at full load operation decreased marginally with an increase of injector opening pressure with test fuels as seen from same table. Improved spray characteristics and reduction of size of fuel particle might be responsible factors to decrease TOPP at full load with an increase of injector opening pressure with test fuels. This once again established the fact by observing drastic increase of peak pressure and lower TOPP at full load operation, engine with LHR combustion chamber showed improved performance over CE.

SUMMARY

- Engine with LHR combustion chamber is efficient for alternative fuel like biodiesel rather than neat diesel.
- Engine with LHR combustion chamber with biodiesel improved its performance over CE at recommended injection timing and optimized timing.
- The performance of the engine improved with advanced injection timing, increase of injector opening pressure and with preheating with both versions of the combustion chamber with biodiesel.

Novelty

Engine parameters (injection timing and injection pressure) fuel operating conditions (normal temperature and preheated temperature) and different configurations of the engine (conventional engine and engine with LHR combustion chamber) were used simultaneously to improve performance and combustion characteristics of the engine. Change of injection timing was accomplished by inserting copper shims between pump frame and engine body.

Highlights

- Fuel injection pressure & timings affect engine performance.
- Performance improved with preheating of biodiesel
- Change of combustion chamber design improved the performance of the engine

Future Scope of Work

The performance of both versions of the engine can further be improved by preheating of biodiesel

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